2-Stroke Engine Options for Automotive Use: A Fundamental Comparison of Different Potential Scavenging Arrangements for Medium-Duty Truck Applications

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Abstract

The work presented here seeks to compare different means of providing scavenging systems for an automotive 2-stroke engine. It follows on from previous work solely investigating uniflow scavenging systems, and aims to provide context for the results discovered there as well as to assess the benefits of a new scavenging system: the reverse-uniflow sleeve-valve.

For the study the general performance of the engine was taken to be suitable to power a medium-duty truck, and all of the concepts discussed here were compared in terms of indicated fuel consumption for the same cylinder swept volume using a one-dimensional engine simulation package. In order to investigate the sleeve-valve designs layout drawings and analysis of the Rolls-Royce Crecy-type sleeve had to be undertaken.

A new methodology for optimization was developed and the analysis process also took into account work done by the charging system, this being assumed to be a combination of supercharger and turbocharger to permit some exhaust waste heat recovery.

As a result of this work it was found that the opposed-piston configuration provides the best attributes since it allows maximum expansion and minimum heat transfer. It gave net specific fuel consumption results which were 9.6% lower than the loop-scavenged engine (which was marginally the worst of the configurations investigated). The other uniflow systems were next, with the reverse sleeve valve being the most promising (3.4% better than the loop-scavenged engine).

Furthermore, although the general performance the loop-scavenged configuration was closer to the other designs than was initially expected, it was found to be compromised by its requirement to have intake and exhaust ports at the same height in the cylinder, thus lengthening the gas exchange events for any given angle-area and consequently reducing the effective (or trapped) compression and expansion ratios. This was despite the use of a charge trapping valve to provide asymmetric port timing and minimize charge short-circuiting, the adoption of which was felt to be a factor in its better-than-expected performance. Finally, the reverse-loop-scavenged poppet-valve type was found to be so compromised by breathing and valve train kinematics that it was not taken to a full optimization.

For the opposed-piston engine, once the port timing obtained by the optimizer had been established, a supplementary study was conducted looking at the effect of relative phasing of the crankshafts on performance and economy. This was found to have a small effect on fuel consumption for a significant change in compression ratio, suggesting that, if available, variable crankshaft phasing could be a very important control actuator for gasoline compression ignition in such an engine.

Importantly, it was found that existing experiential guidelines for port angle-area specification for loop-scavenged, piston-ported engines using crankcase compression could also be applied to all of the other scavenging types, this having been done here in order to provide a starting point for the work. This important result has not been demonstrated before for such a wide range of architectures. The optimizer employed then allowed further improvements to be made over the starting point. The paper therefore presents a fundamental comparison of scavenging systems using a new approach, providing insights and information which have not been shown before.

Introduction

Context and the need for greater investment in combustion engine technology

Automotive transportation has helped to revolutionize society and thus in many ways the internal combustion engine (ICE) helped to define the twentieth century. However, the synergistic development of the ICE with fossil fuels has resulted in the emission of large quantities of carbon dioxide which, because it is a greenhouse gas, contributes to global warming. It is imperative that all measures are taken to reduce the fossil CO_2 impact of transportation while still providing the economic benefits of affordable transportation that the ICE has brought. While many commentators consider that future ground transportation should consequently be provided solely by battery electric vehicles and the hydrogen proton exchange membrane (PEM) fuel cell (FC), due to the opportunity that these provide to decarbonize their energy supply through renewable electricity generation, there is considerable belief within industry and academia that due to the cost involved – both in terms of infrastructure and to the vehicle purchaser – this would take many years to complete.

Conversely, there is unarguably considerable potential left in the internal combustion engine (ICE) with regards to improving its efficiency. This statement is made simply because whereas the efficiency of an electric motor is in the region of 93-97% across the majority of its operating map, the *peak* thermal efficiency of a typical passenger car or medium truck combustion engine is only in the region of 37-45%. This means that logically there is a greater potential to reduce losses in the ICE, even though it is limited by the efficiency of the Carnot cycle whereas the electric motor and fuel cell (FC) are not. Even though the energy storage system for an ICE boasts an efficiency of 100% (disregarding its minimal evaporative emissions) – versus about 85% for a battery – in terms of energetic efficiency the ICE system cannot get back on par with that of an electric vehicle (EV). On the other hand, the amount of energy (and with it the achievable range of the vehicle) that can easily be stored in hydrocarbon form on board a vehicle dwarfs that which a battery can currently hold, and this state of affairs will continue until some significant breakthroughs in battery chemistry are made. These, if they are ever realized, will then take of the order of two decades to bring to mass production.

A further issue is that whereas the ICE has historically been proven to be a silver bullet for all forms of transportation, alternative propulsion systems do not have the same potential reach. Indeed, heavier means of surface transportation such as long-distance haulage and shipping may never be able to adopt these solutions (mainly due to the challenges of energy storage). This is even more likely for long-distance commercial aviation, despite efforts being made to produce battery-electric and PEM FC-powered light aircraft.

Against this backdrop it is obvious that technologies that improve the efficiency of all forms of the ICE ought to be seen to be of crucial importance for the foreseeable future. This is true regardless of the type or degree of electrical hybrization that may be applied to it. When it is further considered that the vehicle original equipment manufacturers (OEMs) will undoubtedly require a portfolio approach to comply with emissions legislation during any interim period of migration to a fully-electric future, improvements to the ICE will also ensure their continued relevance for an ever-longer time period. Due to their incomparable cost-effectiveness for all stakeholders in the transportation economic system, this in turn strengthens the business case for investing in ICE technology.

The 2-stroke engine in transportation

In the automotive world the reciprocating 4-stroke engine has utterly dominated all other forms of prime mover to date. The reality is that it was the invention of this type of engine by Nikolaus Otto, together with its utility, which gave rise to the automobile (and not the other way round). In comparison, except for the essentially separate markets of cost-effective and high-performance motorcycles, the 2-stroke engine has historically been neglected for light- and medium-duty applications (although it must be remembered that numerous cheap cars were built using very simple 2-stroke engines in Eastern Europe until the 1980s). In the West even the rotary engine was more successful than the reciprocating 2-stroke engine in passenger cars, with Mazda producing significant (if still small in comparison to their 4-stroke engine volumes) numbers of Wankel-type rotary engines until production of that line of vehicles ceased with the RX-8 in 2012.

This is the case in the automotive world. However, in areas where either power density or efficiency were the primary goals for a particular engine application the 2-stroke cycle reigned, and continues to reign, supreme: the largest and smallest reciprocating engines operate on this cycle. Considering the specific types (not the production volumes) of engines designed and developed for all vehicular applications, one might observe that in fact the 4-stroke engine is in fact an automotive peculiarity.

Sher, in reviewing 2-stroke scavenging, notes that all engines prior to Nikolaus Otto's 4-stroke device were 2-strokes [1]. Sir Dugald Clerk patented what may be considered the first commercially successful 2-stroke engine in 1881. This was what would now be termed a reverse-uniflow engine, utilizing a one-way inlet valve in the head and piston-controlled exhaust ports [1]. Sir Harry Ricardo's later Dolphin engine was similar to this, albeit with a slightly different valving mechanism. It was Joseph Day who, together with one of his workmen, Frederick Cook, developed the piston-ported 2-stroke engine in Bath in 1889-1891 [2]. Allegedly, and like the designs of Clerk, this was to circumvent the Otto 4-stroke engine patents of 1876. Notwithstanding this, the 2-stroke cycle was not embraced with enthusiasm by the automotive industry; one can imagine that this was because the Otto cycle was much simpler to comprehend and optimize within the understanding of engines at that time because of its meaningful separation of cycle events. This situation is true whether the engine employs spark-ignition (SI) or compression-ignition (CI) combustion.

As the engineering science pertaining to the thermodynamics of combustion engines has developed it has become apparent that the 2-stroke cycle does in fact possess some very significant benefits, especially versus the simple throttled SI 4-stroke engine. These advantages can be summarized as:

1. The minimization of pumping work, through the elimination of a dedicated induction stroke. Instead of the induction process being undertaken at the same expansion ratio as the combustion part of the cycle, the 2-stroke is free to adopt scavenge arrangements more-optimized for this purpose and so mitigate this loss.

2. A potential for reduced friction. This is mainly due to two things: the potential for a simpler mechanism for the gas exchange process (through the removal of a requirement for a half-engine-speed drive for this purpose) and increased mechanical efficiency, because

   $$\eta_{Mechanical} = \frac{BMEP}{IMEP} \qquad \textit{Eqn 1}$$

   Where $\eta_{Mechanical}$ is the mechanical efficiency, *BMEP* is brake mean effective pressure and *IMEP* is the indicated mean effective pressure.

3. The potential for simpler and lighter construction. This is again related to the observations on gas exchange mechanisms, and has ramifications in terms of cost and engine mass. It is also related to the specific power increase possible and the fact that this can be traded off against the BMEP necessary, resulting in a requirement for lower peak cylinder pressures and hence lighter scantlings of the engine.

4. Related to 3. above, for a given swept volume, at any operating condition the BMEP required of the 2-stroke engine is half that its 4-stroke counterpart. This reduced in-cylinder load leads to reduced peak cycle temperatures and hence lower emissions of oxides of nitrogen (NOx). For similar reasons the thermal losses are lower.

For these reasons, the 2-stroke engine has been very successful in applications where maximum efficiency has been of overriding importance or where its simplicity and power density advantages can be realized within the prevailing emissions legislation (for example, in marine propulsion, where NOx limits are relatively high, and portable equipment, where cost is an over-riding imperative, respectively). Although it has never achieved significant impact in the automotive market, it has periodically been investigated as an alternative to the 4-stroke engine for on-road applications; for example, many OEMs researched piston-ported 2-strokes in the 1980s as a result of work undertaken by Orbital Engine Corporation [3,4] and Ricardo and Toyota both investigated poppet-valve reverse-loop-scavenged engines at a similar time [5,6], although no production applications were forthcoming. More recently Lotus proposed and demonstrated a variable compression ratio (VCR) loop-scavenged engine which could be started and run in homogeneous-charge compression ignition (HCCI) on a wide range of fuels and with no assistance from spark ignition [7,8], and the Achates Power have helped to promote interest in the opposed-piston engine for medium- to heavy-duty on-highway applications [9,10,11,12].

The disadvantages of the 2-stroke cycle primarily stem from the lack of a dedicated exhaust stroke making scavenging of the burnt gases and their replacement with fresh charge problematic. It is not often realized that a 4-stroke cycle engine actually swaps functions every other revolution of the crankshaft: half of its life it is a combustion engine and the other half it is a very heavily-constructed scavenge pump. A 2-stroke uses its crank-rod-slider mechanism as an engine continuously, but needs a means external to the combustion chamber to achieve gas exchange. This is true whether it uses the underside of the piston or a separate pump (so-called crankcase or external scavenging arrangements respectively).

Functionally, because the exhaust and intake phases necessarily overlap to a degree, there is considerable potential for charge short-circuiting. For a simple crankcase-scavenged piston-ported engine with mixture formation outside of the engine the unburned fuel loss is significant. This leads to high unburned hydrocarbon (UHC) emissions in the exhaust and also to a significant increase in fuel consumption. This works counter to the theoretical advantages of the 2-stroke engine discussed above.

Because of the likelihood of charge air loss direct from the cylinder, achieving lambda = 1 in the exhaust is very difficult. This being the case, the use of a simple three-way catalyst to control emissions (as used in conventional 4-stroke SI engines) is impossible. This is the reason why in this work gasoline compression ignition is used as the combustion mode, as will be discussed later.

The way in which throttling losses are reduced through the removal of the 4-stroke engine's positive scavenging strokes means that at part load the 2-stroke engine necessarily operates with large quantities of residual gas. For SI applications this gives rise to poor combustion, which worsens as the load (and with it the amount of air flowing into the cylinder and available to displace the burnt gas) decreases. Eventually some cycles completely fail to ignite because they are below the ignitability limit, in turn meaning that the next one will have a higher proportion of fresh unburned charge, which then permits combustion initiation. The engine alternately fires and is then said to be '4-stroking'. This is not a term of endearment, since UHC emissions and fuel consumption both then deteriorate further.

Diesel applications do not suffer in the same way but, especially for loop-scavenged engines, without the structured air motion resulting from the positive piston motion in the pumping strokes there are challenges with fuel-air mixing. The large quantity of residual gas that is trapped, however, does mean that the 2-stroke is a natural arena in which to apply HCCI-type combustion systems, and this has historically been successfully demonstrated in a variety of such engines by Honda and others [13,14,15]. The use of these types of combustion system, together with direct injection (DI), therefore provide a feasible route to low pollutant emissions and high efficiency in the 2-stroke engine. This work seeks to investigate the efficiency levels that can be achieved when different scavenging mechanisms are used for the 2-stroke engine when it operates on just such a combustion system.

Before describing the individual scavenging systems analyzed, a short description of the Burt-McCollum sleeve valve (as used in the Rolls-Royce Crecy) will be made, since these mechanisms have not been used in mass-production engines for over 50 years and this will aid in understanding two of the systems investigated here.

The Burt-McCollum Sleeve Valve

A sleeve valve is a form of gas exchange mechanism which fits slideably as a sleeve between the cylinder wall and the piston. Two main types of such valves have been productionized: the Knight double sleeve and the Burt-McCollum single (or mono) sleeve.

The Knight double-sleeve mechanism uses two concentric sleeves which reciprocate, each driven by a crank-rod-slider-type mechanism at half engine speed (for the 4-stroke engines that they were exclusively used in in production). Ports in the top of the sleeves enabled operation on the 4-stroke cycle due to the way in which the sleeves were timed and how the ports interacted with each other. These engines were reportedly very quiet in operation, but were costly to produce and consumed oil at a rate markedly above then-current engines with poppet valves [16]. They were also problematic to start when cold: a corollary of all of the sliding surfaces they had and the fact that they all merely reciprocated along a single line. For these reasons they were not considered successful.

The Burt-McCollum[1] sleeve valve, while at first glance similar, is fundamentally different. Here the single sleeve moves in an elliptical path, providing two dimensions in which ports in the sleeve can align with intake and exhaust ports in a 4-stroke engine. A representation of a Burt-McCollum sleeve is shown in Figure 1. Conventionally the sleeve was driven in this elliptical path by a peg which was mounted on a half-engine-speed gear; a hole in a drive ball, itself held in a socket carried in an extension to the sleeve base, allowed the peg to slide within it and the resulting motion to become elliptical as the sleeve moved in an arc as it simultaneously reciprocated. Figure 1 shows the extension to the sleeve that the sleeve ball carrier shown in Figure 2 bolted to in the case of the Bristol Centaurus engine [17].

[1] The Burt-McCollum single sleeve valve contains elements that were separately invented by Peter Burt, a Scottish inventor and designer of engines for Argyll, and James McCollum, a Canadian engineer. In the form that it was put into mass production it arguably owes more to the former than the latter, and hence, possibly, the order of their names adopted in describing it [17].

Fig. 1: CAD model of Burt-McCollum sleeve as used in the Bristol Centaurus aero engine. Reproduced from [17].

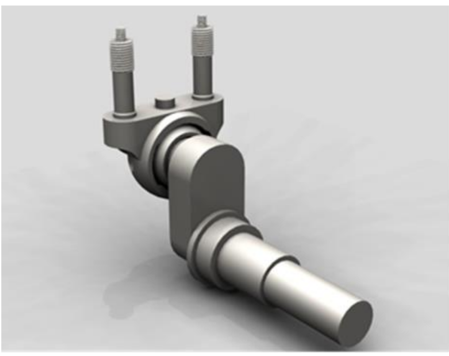


Fig. 2: CAD model of ball-and socket joint arrangement used in the Burt-McCollum sleeve as used in the Bristol Centaurus aero engine. The sleeve drive peg is on the crank that was in turn driven by a half-engine-speed gear in this engine. Reproduced from [17].

Figure 3 shows a representation of a cylinder set from the Bristol Centaurus 4-stroke engine, whose design was used for prior work investigating the angle-area potential of a sleeve valve versus a four-poppet-valve-per-cylinder arrangement [17]. This previous study showed that even applying the latest limits on valve sizes and lift profiles, the single-sleeve valve offers significant extra breathing potential over the current automotive state of the art. Figure 3 shows the sleeve towards the top of its travel, indicating that the piston is at top dead centre (TDC) firing.

Fig. 3: CAD model of the crankshaft and a cylinder set from the Bristol Centaurus aero engine which used Burt-McCollum sleeve valves for gas exchange. Reproduced from [17].

Because it had fewer parts the Burt-McCollum sleeve valve was much cheaper to produce than the Knight arrangement. Famous engines to use the single-sleeve valve include many of the Bristol air-cooled radial aero engines and the Napier Sabre. The Centaurus, an 18-cylinder twin-row radial engine and at 53.62 litres the largest of the Bristol reciprocating aero engines, reportedly had the longest time between overhauls of any piston aero engine, and once external sleeve-contacting rings (which were stationary in grooves in the cylinder wall and acted inwards on the base of the sleeve) were adopted and the importance of oil temperature control was realized it reportedly had extremely low oil consumption too [18].

In addition to being desmodromic mechanisms, both Knight and Burt-McCollum valves conventionally employed what amounts to a fixed piston as a cylinder head; this is termed a junk head and is visible in Figure 3. It carries junk rings – stationary piston rings – to permit sealing of the top of the combustion chamber. Because there are no poppet valves, the combustion chamber can be ideally shaped and cooled; there was ample room for the twin spark plugs necessary in the aero engines that such valves were very successfully applied to, for instance.

Sir Harry Ricardo was a major proponent of single-sleeve valves, and in addition to supporting Bristol, Napier and Rolls-Royce in their 4-stroke applications he had experimented with their application in 2-stroke engines [19]. The specific embodiment championed by Ricardo was to use the sleeve as a control to permit asymmetric timing of a uniflow scavenging system, with the exhaust port at the top and intake at the bottom. Here he took things considerably further and applied stratified charge fueling as well, with the resulting combustion system being able to operate unthrottled over large portions of the operating map. Both Rolls-Royce and Napier built test engines but with the latter occupied in developing the Sabre engine it was Rolls-Royce that received a development contract for what became the Crecy, a 26.1 litre 90° V12 2-stroke engine intended for high-speed interceptor aircraft applications. A sectional drawing of the Crecy is shown in Figure 4 [20], and its operating cycle is depicted in Figure 5 [21] with approximate event timings after and before top dead centre (ATDC and BTDC, respectively); in Figure 5 the arrangement of the fuel injector and twin spark plugs in the combustion chamber bulb can readily be discerned.

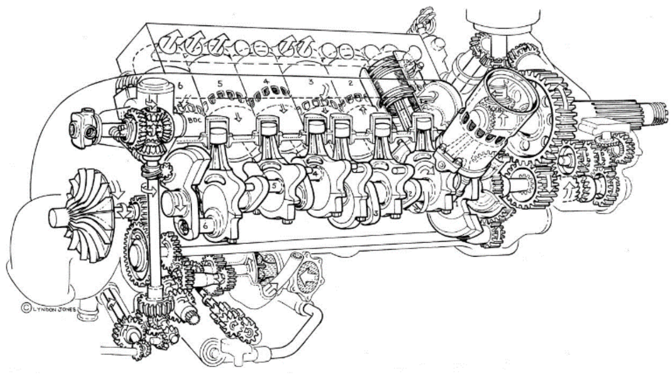
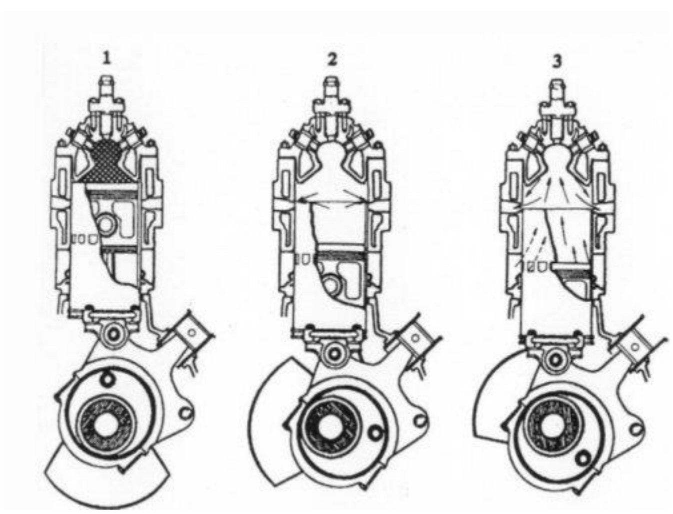


Fig. 4: Sectional drawing of the Rolls-Royce Crecy sleeve-valve 2-stroke engine (in its original non-compounded form). Reproduced from [20], courtesy of The Rolls-Royce Heritage Trust; copyright the estate of Lyndon Jones.



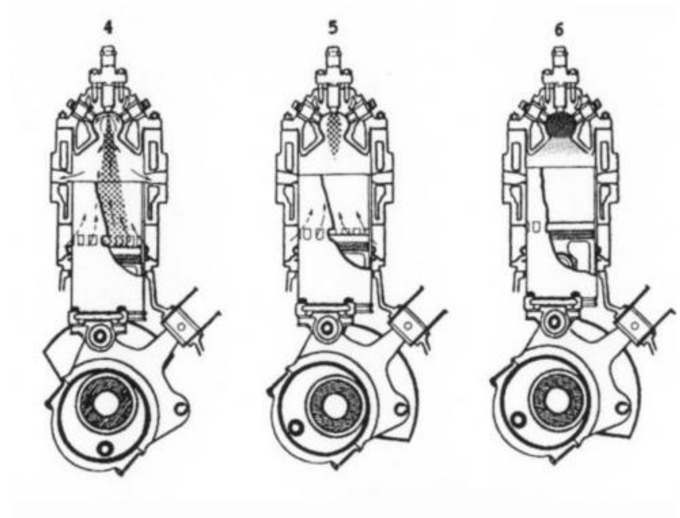


Fig. 5: Rolls-Royce Crecy operating cycle. 1: combustion (0° ATDC); 2: end of expansion with exhaust port at top of sleeve valve about to open (start of blowdown) (~90° ATDC); 3: exhaust and intake ports open – note upwards uniflow scavenging flow (~135° ATDC); 4: exhaust ports starting to close (end of scavenging), start of fuel injection (180° ATDC); 5: end of fuel injection, supercharging via intake ports (~°135 BTDC); 6; start of compression, with rich mixture trapped in combustion chamber bulb (~120° BTDC). Reproduced by permission of Rolls-Royce Heritage Trust from [21], copyright Rolls-Royce plc.

Inspection of Figures 4 and 5 readily shows that the Crecy employed an elegant form of sleeve drive via yoke plates driven directly by eccentrics on the crankshaft (this being possible since it was a 2-stroke engine). These had the drive pegs attached to them, and each yoke drove two sleeves. A dummy piston was used to stop yoke rotation. This mechanism still gave rise to an elliptical motion to the sleeve. A final mechanical point of interest is that while the intake ports in the sleeve were necessarily composed of windows, the exhaust was of a '360°' configuration, i.e. it was pulled clear of the junk head in its entirety with no windows. This gave the minimum duration for the necessary angle-area but also meant that blowby into the void above the sleeve and then out into the exhaust port would have been considerable. This will be returned to later.

A representation of the port timing is given in Figure 6; this is from work conducted for this paper in order to give angle-area information for the one-dimensional (1-D) analysis work conducted here. This graphical construction was undertaken from information given in [21].

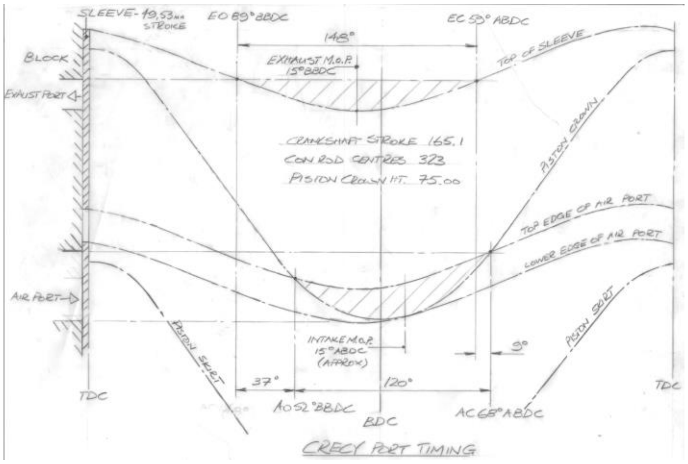


Fig. 6: Example of port timing drawing development undertaken for the Rolls-Royce Crecy for the work reported here.

Finally, after World War 2 the sleeve-valve 2-stroke engine was further investigated by S.S. Tresilian for Rolls-Royce as a major component in a highly-integrated aircraft propulsion system. This was the so-called 'X-engine' [21] which sought to maximize the architectural possibilities of the sleeve valve within a 16-cylinder 4-row in-line radial engine, ultimately to have been of approximately 9.0 litres swept volume and employing high levels of turbocompounding. It did not progress as a concept, because of the rapid improvement of the gas turbine occurring at that time. The subject of turbocompounding, and which of the different engine types discussed in the present work might suit it best, is returned to later.

For more detailed information on sleeve valves and their application see [20,21,22,23,24,25].

Scavenging systems investigated

The work presented here seeks to compare six different scavenging systems for a 2-stroke engine suitable to power a US light-duty truck. The scavenging system effectively defines the major architecture of a 2-stroke engine, and there are several fundamental types that can be utilized. Together with the combustion system it dictates the performance and fuel consumption of the engine. Sher provides an excellent overview of some scavenging types and their history [1]. The way in which the comparison was made is described later. The scavenging systems investigated here were:

1. The opposed-piston 2-stroke (OP2S) engine, which has successfully been applied to aircraft propulsion as well as to engines for power generation and rail traction and has recently been promoted by Achates Power for heavy-duty applications [9,10,11,12,20,26,27,28].

2. The poppet-valve uniflow configuration, as exemplified by the Detroit Diesel 2-stroke engine [29] and various marine engines [30,31,32] and which is being reevaluated now for automotive use [33]. This is referred to herein as the 'port-poppet' type.

3. The sleeve-valve uniflow arrangement, which was used in the Rolls-Royce Crecy described in the previous section. In this work this will be referred to as the 'forward-uniflow sleeve' [21].

4. The reverse-uniflow sleeve-valve engine. This is based on the Crecy arrangement and is referred to here as the 'reverse-uniflow sleeve'. Conceptually it might be considered to be a hybrid of the Crecy and Clerk's engine of 1881. It was analyzed partly due to the findings of an earlier paper which compared it to the OP2S and port-poppet engines [34].

5. The reverse-loop poppet-valve engine, as previously proposed by Toyota and Ricardo [5,6] and currently being investigated by Renault and others [14]. This shares the most clear architectural links with the current automotive norm, the poppet-valve 4-stroke engine. This type was introduced to give a more complete comparison using technology in production now.

6. The Schnürle-loop scavenged engine, referred to in this work as 'loop-scavenged'[2]. This is thought of by many as the classical 2-stroke engine configuration, in the same way that the poppet-valve 4-stroke engine is now considered a norm [1,35].

These arrangements are shown schematically in Figures 7 to 12.

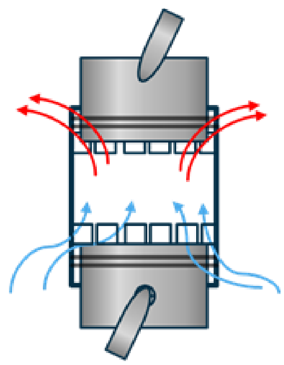


Fig. 7: Schematic representation of the opposed-piston engine (OP2S). In this diagram the exhaust piston and ports are at the top and the inlet piston and ports are at the bottom. Port timing is controlled by the pistons, which do not have to be in phase, making asymmetric port timing possible. The primary scavenging air motion is upwards.

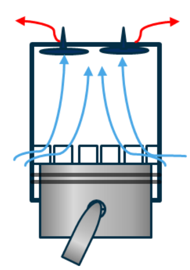


Fig. 8: Schematic representation of the poppet-valve uniflow configuration. In this diagram the exhaust valves and ports are at the top and the inlet ports

[2] Sher [1] describes the true Schnürle-loop scavenging as being only by intake ports to the sides of the exhaust port. He refers to the evolved configuration which most now consider to be Schnürle-loop, having intake ports opposite the exhaust port as well, as being 'Curtis-type'. Here the term Schnürle-loop will be used exclusively to encompass this evolution as well.

are at the bottom and uncovered by the piston. Asymmetric timing is obviously possible. The primary scavenging air motion is upwards.

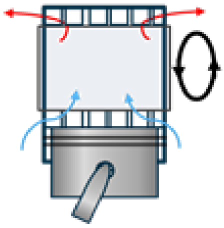


Fig. 9: Schematic representation of the forward-uniflow sleeve-valve arrangement, as used in the Rolls-Royce Crecy. In this diagram the exhaust ports are at the top and the inlet ports are at the bottom. Both sets of ports are uncovered by the sleeve which moves elliptically. Asymmetric timing is possible. The primary scavenging air motion is upwards.

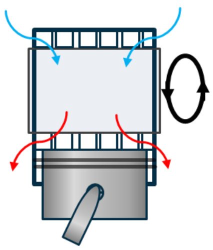


Fig. 10: Schematic representation of the reverse-uniflow sleeve-valve scavenging arrangement. In this diagram the inlet ports are at the top, and the exhaust ports are at the bottom. The ports are uncovered by the sleeve which moves elliptically. Asymmetric timing is possible. The primary scavenging air motion is downwards.

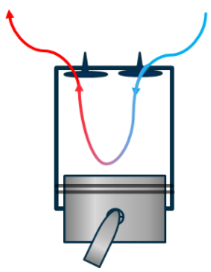


Fig. 11: Schematic representation of the reverse-loop-scavenged poppet-valve engine. Intake and exhaust valves are situated at the top of the cylinder as per the current automotive norm. In this configuration asymmetric timing is possible using conventional cam phasing devices. The primary scavenging air motion is down, across the piston crown and up, with this being heavily assisted by the ports' angles to the cylinder; the intake is conventionally vertical and situated between the camshafts.

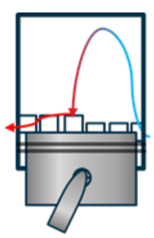


Fig. 12: Schematic representation of the Schnürle-loop scavenged engine. Both the exhaust and intake ports are positioned at the base of the cylinder and their primary timing is controlled by the piston. In this configuration asymmetric port timing is not possible without further complication being added (see text). The primary scavenging air motion is up, across the combustion chamber and down, with this being assisted heavily by the ports' angles to the cylinder.

The Schnürle-loop scavenged engine has historically had various complications applied to it to improve its efficiency, with a valve in the exhaust similar to the 'YPVS' system used by Yamaha being very common. Such valves vary the exhaust port height non-cyclically, and so affect exhaust port opening (IPO) and closing (IPC) timing equally. Since the Schnürle-loop scavenged configuration represents what many consider to be the conventionally-conceived 2-stroke engine, it represents something of a baseline for the other arrangements. It was not assessed as part of the previously-reported work [34] and so it was included here. However, in including it it was also decided to assess the most flexible of the exhaust timing adjustment mechanisms applied to this type, which was considered to be the Lotus charge trapping valve (CTV) described by Blundell and co-workers in several engine research projects [7,8,15,36]. This would set a 'best case' baseline to compare the other scavenging systems to[3].

Configurations (1), (2) and (3) were described in a previous paper in which only uniflow scavenging arrangements were compared [34]. In that work the OP2S engine emerged as a clear winner, with the port-poppet configuration second but only marginally better than the forward-uniflow type. In [34] the observation was made that the port-poppet type was severely restricted by valve train dynamics and as a consequence, since it is desmodromic and is not limited by such issues, it may well be that the forward-uniflow-sleeve might be superior under some situations. In the analysis of the angle-areas undertaken for the Crecy for the earlier work, it was determined that the flexibility provided by the sleeve's port layout could permit additional optimization possibilities. Furthermore, with the scavenge air flowing in the reverse direction (i.e., from the top down) in this new layout, the combustion chamber and the top of the sleeve could be expected to run cooler, although it was accepted that the exhaust leaving the base of the cylinder would likely mean that the piston would then run hotter (a common challenge for the exhaust piston of the OP2S configuration as well). A cooler-running combustion chamber might be expected to extend the knock limit in SI applications. As a consequence, configuration (4) has been included in the present work to assess whether it has merits in comparison to the OP2S.

All of the concepts were compared in terms of net indicated fuel consumption for the same cylinder swept volume of 751 cc, being an individual cylinder swept volume suitable for a medium-to-heavy duty engine (i.e. the sectors where it is expected that such high efficiency 2-stroke engines will be introduced first). Similarly a geometric compression ratio (CR) of 15:1 was adopted for each. However, since the ratio of stroke length to bore diameter cannot be kept constant for all of the arrangements without severely compromising one or more of them, this was altered to better suit their requirements; for example, the OP2S engine used the same total stroke:bore ratio of 2.2 as Achates [11,12]. Once these were set, no further individual optimization of this variable was conducted. It was

[3] Schnürle-loop scavenging represents a clear advantage over earlier piston-ported single-piston designs such as cross- or loop-scavenging [1]. The latter are rarely used now. The required deflector piston detail for cross scavenging compromises the combustion chamber shape, and in the conventional loop-scavenged engine the position of the exhaust port above the transfer means charge trapping is poor.

also assumed that a 4-stroke-style wet crankcase design would be used, i.e. that all engines would use external scavenging (see below). Operationally, the gasoline compression ignition (GCI) combustion system that was modelled for all variants used the same combustion profile for each of the operating points investigated (see later), this being the result of previous work conducted by Aramco [37]. This combustion system was adopted both to simplify the modelling process and because it provides very low engine-out NOx, which will be of crucial importance to 2-stroke operation where it is difficult to provide the lambda = 1 conditions in the exhaust necessary for a three-way catalyst to work properly.

The engine specifications used in this work are listed in Table 1.

Table 1. Engine specifications modelled.

Engine Type	**Opposed-Piston (OP2S)**	**Port-Poppet**	**Sleeve-Valve Forward Uniflow**	**Sleeve-Valve Reverse Uniflow**	**Poppet-Valve Reverse Loop**	**Piston-Ported Loop (Schnürle)**
Basic Scavenging System	Uniflow	Uniflow	Uniflow	Uniflow	Loop	Loop
Bore [mm]	75.75	86	86	86	98.52	98.52
Stroke [mm]	166.65	129.29	129.29	129.29	98.52	98.52
Stroke:Bore Ratio [-]	2.2	1.5	1.5	1.5	1	1
Conrod Length [mm]	166.65	258.58	258.58	258.58	197.04	197.04
Cylinder Surface Area difference [%]	+4.56	0	0	0	-1.73	-1.73

Note that for the OP2S engine, the terms TDC and bottom dead centre (BDC) are referenced from the exhaust piston angular position; typically the exhaust piston leads the intake.

Before describing the modelling methodology adopted, it should be noted that no attempt was made to specify and model an equivalent 4-stroke engine. This was due to the complexity of the undertaking and its difference to the majority of the engines. This was considered acceptable because other researchers have already done this for some scavenging configurations. Specifically, in [38] it was estimated that a OP2S diesel engine would have 13-15% better fuel consumption than its 4-stroke equivalent, and later work by those authors suggested that a OP2S would still have 7% improvement and be broadly equivalent in fuel consumption terms to a compounded 4-stroke engine [39]. This will be returned to later.

Description of the simulation method

In this paper we focus on the simulation results for lowest fuel consumption, and all of the results quoted are given on an indicated basis. This is because the construction of full engine models and the estimation of friction losses was outside the scope of the current level of work; however it is recognized that sleeve valves would likely have different friction and heat loss behaviours.

Three engine operating points were used to compare the different scavenging system designs. These were:

A 1500 rpm, 3 bar IMEP, 1.2 bar exhaust manifold pressure

B 1500 rpm, 14 bar IMEP, 2.0 bar exhaust manifold pressure

C 3000 rpm, 12 bar IMEP, 2.5 bar exhaust manifold pressure

These operating points took into account the medium-duty nature of the study: point A was intended to be a representative part-load operating point, and B and C were notionally peak torque and peak power respectively; they will be referred to in this manner going forward. Points B and C turn represent nominal indicated specific outputs of 225 Nm/l and 60 kW/l for this cylinder capacity, i.e. reasonable performance for this type of application.

Each model was created as a generic single-cylinder version of the specific concept using GT-Power, a 1-D engine simulation software package. The models consisted of the cylinder and porting arrangements, with user-imposed conditions either side via a series of 0-D elements. The OP2S engine was modelled as an equivalent single cylinder where the piston area was doubled and cylinder head area set to zero to represent the opposed piston arrangement. The equivalent single piston motion was calculated to account for the relative motion of the two pistons. For all models, the direct fuel injection quantity was calculated from the desired air-fuel ratio (AFR). At operating point 1 the AFR was held at 43.7 for all configurations, while at operating points 2 and 3 the AFR was increased to 16.2 for all cases. As mentioned above, these were the result of earlier work by Aramco on GCI [37]; the heat release profile adopted is shown in Figure 13.

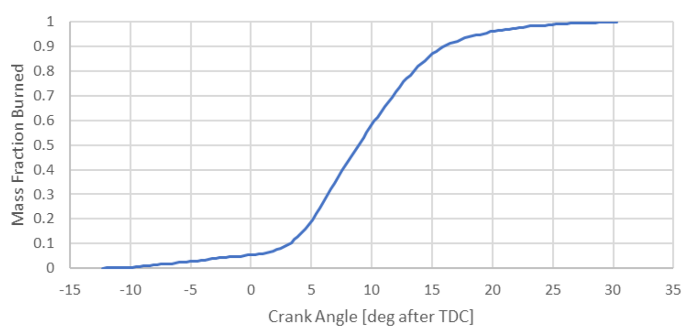


Fig. 13: Combustion heat release profile adopted from work performed by Aramco on GCI [37].

Intake air pressure was controlled to achieve the target IMEP via a closed loop. While the exhaust pressure was user-imposed, exhaust pressure sweeps were also performed to verify trends that were seen at the individual operating points.

To minimize friction and achieve a degree of waste heat recovery scavenging air supply was assumed to be provided by a system comprising turbocharger and supercharger in series, although this was not explicitly modelled. In order to evaluate fuel consumption meaningfully, the power to drive the mechanical supercharger had to be accounted for, and so a net specific fuel consumption (NSFC) was calculated as follows:

$NSFC\ [g/kWh] =$

$$\frac{Fuel\ Flow\ [g/h]}{(Indcated\ Power\ [kW] + Supercharger\ Drive\ Power\ [kW])}$$

Eqn 2

Supercharger drive power was estimated from an energy balance of the exhaust and intake conditions, with the difference between what was required and what the turbocharger could supply assumed to be provided by the supercharger. In order to remove the effect of changing assumptions, all turbomachinery efficiencies were assumed to be 70% in this work.

Since no detailed in-cylinder flow modelling was conducted, the scavenging behavior of each design was dictated by a profile which relates in-cylinder to exhaust manifold burned gas fractions. This was determined from an extensive survey of the literature and changes between scavenging types. These profiles are shown in Figure 14; the scavenge profile for the OP2S was taken from work by Mattarelli *et al.* [40], for the port-poppet and both sleeve valve arrangements from a uniflow study described by Laget *et al.* [41], and for the reverse-loop poppet-valve it was adapted from the Renault Powerful concept [42].

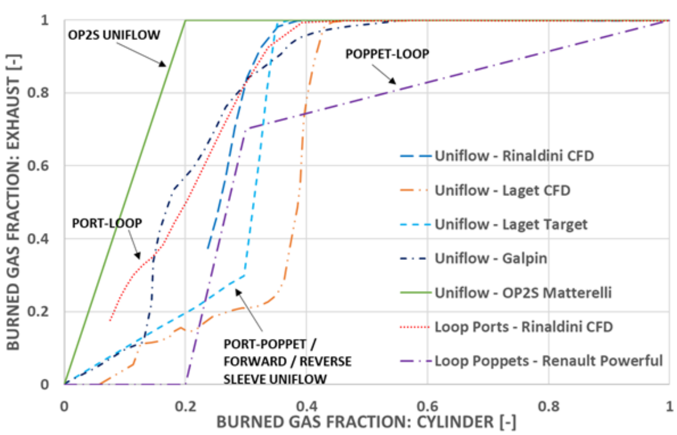


Fig. 14: Scavenge profiles used in the models [40,41,42], together with other example profiles taken from literature.

When analyzing the port and valve optimization results specific time-area was used extensively. This metric has historically been used to assist in the design of 2-stroke engine ports and was used here as a guide. The specific time area is defined to be the integral of port open area with time divided by the swept volume. It provides a measure of port availability for gas flow during the cycle and there are different values which therefore affect the performance of an engine:

1. Intake-specific time area – the intake open area calculated over the time interval from intake port opening (IPO) to intake port closing (IPC)

2. Blowdown-specific time area – the exhaust open area calculated over the time interval from exhaust port opening (EPO) to IPO (this is sometimes referred to as the free exhaust period)

As well as the above established metrics, an additional one was used to help gain insight into the scavenging process:

3. Scavenge-specific time area – the minimum of the exhaust and intake port open areas over the interval EPO to exhaust port closing (EPC)

The guidelines recommended by Naitoh and Nomura were used to establish angle-area targets to lead the design of the port geometry at the beginning of the study [43]. These were originally derived for high-performance Schnürle loop-scavenged engines for motorcycle racing, but nevertheless it was believed by the authors that they should applicable to any 2-stroke scavenging configuration, and that they could be used to get sufficiently close to the eventual configuration that a numerical optimizer could then be used (see later). It is believed that this is the first time such an approach has been taken across such a broad range of engine layouts. Although not tested here, it is further believed that this should be the case regardless of whether it crankcase scavenging or an external scavenge pump is employed.

The general constraints on the engine operating envelope for optimization were logically set to be that under normal operation EPO should be before IPO and EPC should be before IPC (i.e. asymmetric timing). As discussed above it was assumed that to supply air to the engine some form of compound charging system would be required, and the work necessary to drive the supercharger component was calculated and applied so that this requirement was accounted for. To reflect this a further constraint was imposed, so as to ensure that there would be sufficient exhaust pressure available to drive a turbocharger in such a system: the pressure in the volume immediately downstream of the exhaust port was set to 1.2 bar at the part-load operating point, to 2.0 bar at peak torque, and to 2.5 bar at peak power (points A, B, and C respectively). These values in turn assume that a meaningful exhaust back pressure downstream of the turbine for a catalyst and silencer system has been catered for.

In terms of process, port timings were determined by numerical optimization of the models at the 1500 rpm 14 bar IMEP maximum torque operating point (Point B)[4]. This was because that while establishing a ranking for minimum in-use fuel consumption was the primary aim, the engines had to meet the performance targets to be viable, and it was felt that the step in air mass flow from peak torque to peak power could be accommodated by changing the boost pressure. NSFC was minimized at this point within the constraints imposed by the geometry and design of each concept. These port timings were then applied at the other two operating points.

Engine porting arrangements

The general port timing arrangements for the OP2S engine are controlled by their respective pistons; the lead of the exhaust over the intake is an important parameter because it is what gives it its asymmetric port timing. A study was conducted to investigate the optimum amount of exhaust piston angular lead over the intake. While it is theoretically possible to vary the timing between the two crankshafts, this was not used as a variable in the present study; once the exhaust lead had been set for Point B it was left fixed for the other operating points investigated.

[4] The term 'port' is used in reference to timing events throughout this paper even when valves and not pistons are used for this purpose.

During the exhaust lead optimization process the injection timing for the OP2S was also compensated to allow for the fact that as the piston phase changes, so the angles of maximum and minimum volume also change, effectively varying the position of TDC and BDC *as far as the engine cycle is concerned* (i.e. the minimum and maximum volume in the cylinder, respectively). Regardless of this, the swept volume was held constant at 751 cc and the geometric CR at 15:1.

In his 1996 book Blair recommended that a maximum port width of 66% could be used to avoid the use of pegged rings [35]. It being expected that improvements in materials might permit an increase, 75% port width open area was adopted here. This was justified because if necessary the rings could be pegged anyway. For a total of 12 ports, this metric gave 22.5° for the subtended angle of each port. Ma *et al.* covered the topic of port widths comprehensively for intake ports on a port-poppet engine recently [33]. For fairness of comparison, this value was adopted for all of configurations using piston-controlled ports, including the sleeve ports for the sleeve-valve engines. However, note that in those cases the respective ports could be larger within the cylinder wall itself since the rings only rub against the internal diameter of the sleeve.

Having set these parameters the main variable left for the cylinder ports was their height. This value affects angle-area and the port timing of the engine simultaneously, and thus the trapped expansion ratio (ER) and the ratio between these two (see below). To maximize work and minimize NSFC one requires maximum ER together with the highest value of the ratio of ER to CR; theoretically if this value reaches greater than unity then one can create a degree of Miller cycle operation. Because of the temptation to perceive a 2-stroke engine as a symmetrically-timed piston-ported device, this is an operating regime that is not normally associated with this type of engine.

While the intake port geometry approach for the port-poppet engine is the same as that for the OP2S, its exhaust process is controlled by cam-driven valves. Since the intention was to maximize expansion then logically the greatest amount of valve curtain area possible would be needed and so in this study four exhaust valves were used for this type. The cam profiles were calculated for typical 4-stroke exhaust valve reciprocating masses and the spring rates selected from an existing 1-D engine simulation model. Scaling rules were used modify them to ensure that valve accelerations and velocities were not exceeded for use in the 2-stroke engine, i.e. that valve control would be maintained over the engine operating range investigated, accommodating the fact that the camshafts are now turning at engine speed. As discussed in previous work on such engines, the limitations on port angle-area imposed by valve kinematics are a factor which limits its performance [34]. This is essentially because the kinematics set a minimum valve event length and the resulting process has to be timed to have the minimum impact on the trapped CR and ER, in turn limiting work extraction as discussed above.

For the forward-uniflow sleeve-valve engine a general engine cylinder scheme was drawn using the dimensions of Crecy. This was then scaled appropriately for the engine being modelled. In Figures 5 and 9 it can be seen that the exhaust exits at the top of the cylinder like the port-poppet engine, but with 360° of port width. While this approach logically minimizes port height for any required angle-area, for the application considered here this was considered impractical because crevice volumes have to be controlled and pressure loss contained, both of which were not considered serious issues in the Crecy. Hence the sleeve was modelled with lands and angles using a similar approach to that used for the OP2S engine; a set of junk rings was assumed to be included to seal the top of the combustion chamber.

At the other end of the cylinder for the forward-uniflow sleeve-valve, the sleeve controls intake angle-area and timing. It can be timed with a lead or lag angle relative to the piston. These were another set of variables that needed to be optimized.

Similar methodologies were applied to both the forward- and reverse-uniflow sleeve-valve engine configurations, but changes in heat transfer were not considered for the two sleeve valve configurations for two reasons: firstly this was primarily a study on gas exchange, and secondly there is conflicting evidence regarding heat transfer being better and not worse for sleeve valve engines [19,21]. Supposedly the beneficial case arises because while it represents an additional barrier, the elliptical nature of the sleeve motion efficiently moves the heat around. During this work it was not considered that sufficient knowledge was available to influence calculations and thus the heat transfer was considered to be similar to that of the port-poppet arrangement for ease of comparison. Further work would be useful to assess this.

For the reverse-loop scavenged engine the valve train kinematics limitations discussed above for the port-poppet engine obviously also apply. However, whereas the port-poppet engine can use its four valves entirely for exhaust angle area, the reverse-loop engine cannot, and its breathing capacity is severely handicapped as a result. Thus there are two significant limitations for this type of engine: kinematics and breathing. To these must be added that charge short-circuiting is also problematic with the exhaust valves/ports juxtaposed with the intakes (unlike in the piston-ported loop-scavenged engine, where they are on the bore periphery), such as in the 'Flagship' engine proposed by Ricardo [5,6]. Placing the intake ports vertically between the camshafts forces those components apart, in turn making the combustion chamber more like a traditional pent roof, but potentially further exacerbating the short-circuiting problem since the ports are then angled towards each other. Ricardo reported compromises with respect to valve sizing, kinematics and scavenging [6], with a suspicion that a severe preignition problem may have been one result of the adoption of the architecture. Benajes *et al.* reported a different layout (i.e. non-pentroof) for a reverse-loop scavenged engine [14], with promising results, but they noted that adopting an early exhaust valve opening (EVO) timing for scavenging purposes reduced expansion and compromised fuel consumption, echoing the results found here. On the positive side for this configuration, twin camshaft phasing devices could readily be adopted to tune intake and exhaust port timing relative to piston motion (assuming the required operation of the valvetrain at crankshaft speed was not a limitation).

After the initial shake down of the models the severe limitations on breathing with this type of engine meant that its performance was sufficiently poor in relation to the other types that, after due consideration, it was not taken further to a full optimization. Hence it is not included in the discussion section that follows. However, this is not to say that as a concept it is entirely without attraction: the ability to share machining lines with existing production 4-stroke engines and the fact that piston rings do not have to traverse ports at all (unlike with all of the other configurations discussed here) are considered to be two major ones, as originally discussed by researchers at Ricardo [5,6]. Nevertheless, in the present research, which was aimed at maximizing efficiency and for which maximum expansion ratio is required, the fact that valve train kinematics force long events which then compromise this aspect of performance was ultimately what led to its being deselected at this point.

Finally, the loop-scavenged piston-ported engine was based on information in the published Lotus 2-stroke papers where a CTV had been employed to modify and tune the exhaust event [7,8,15,36]. The CTV permits asymmetric timing of the exhaust port and can also be variable to permit optimization. The ability to adopt the publicly-available data of Blundell and co-workers for the models was one reason why the stroke-to-bore ratio of this engine was fixed at 1.0, and why it was decided to adopt different values of this parameter for different engine architectures.

Once these engine porting arrangements had been decided upon, a staged approach was taken for the optimization process for the port timing and geometry:

1. The engine models were run at the operating point B (peak torque). Sweeps of the intake and exhaust port timings were then performed.

   Surface response models of the variables of interest (primarily NSFC) were created as functions of the intake and exhaust timing events.

2. An offline optimizer was then applied to these response models to find the optimum whilst adhering to the constraints based on geometry and operating conditions discussed above. The resulting optimal timings were then used to calculate the port/valve geometries required to achieve them.

A schematic diagram of the process followed is shown in Figure 15.

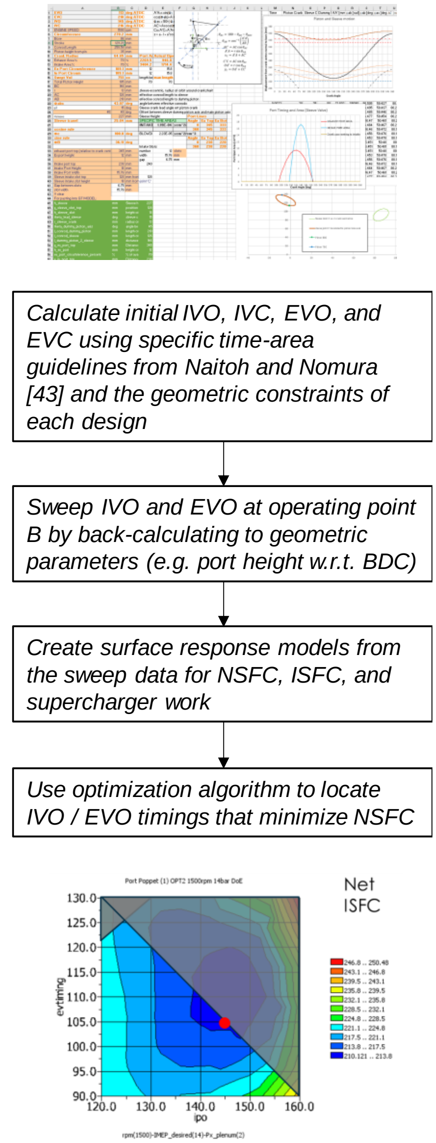


Fig. 15: Schematic diagram of the process of the simulation method used to determine port geometries and timings.

The resulting port geometries and timings were then used to predict the performance at operating points A (part load) and C (maximum power). The NSFC results were then averaged in order to rank the scavenging systems.

Results

The results of the optimization are summarized in Table 2, and the associated port profiles are shown in Figures 16 to 20. Table 3 presents the port timings and Table 4 the associated specific time-area values. For the reasons discussed above, the reverse-loop poppet-valve configuration is not included in any of the results due the indications from early in the project that it would clearly be inferior to the other configurations, meaning that it was not taken forward into the optimization stage.

It is important to note that since the engines have not been modelled in detail the data shown should not be interpreted as absolute values for them, although it can be used for a general comparative ranking of the different approaches.

Table 2: Comparison of results for the different scavenging concepts studied.

Engine Type	NSFC [g/kWh]			Estimated Supercharger Power Requirement [kW]		
Operating Point	A	B	C	A	B	C
OP2S	183	189	192	0.244	0.158	1.96
Port-Poppet	194	211	207	0.210	1.27	3.04
Forward-Uniflow Sleeve	197	214	208	0.224	1.27	3.01
Reverse-Uniflow Sleeve	187	214	201	0.197	2.02	2.68
Piston-Ported Loop	199	214	211	0.230	0.956	3.18

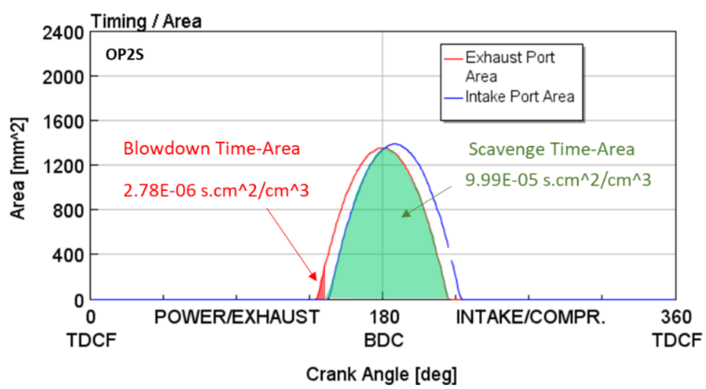


Fig. 16: Port area profiles optimized for lowest NSFC at operating point B (peak torque) for the OP2S engine.

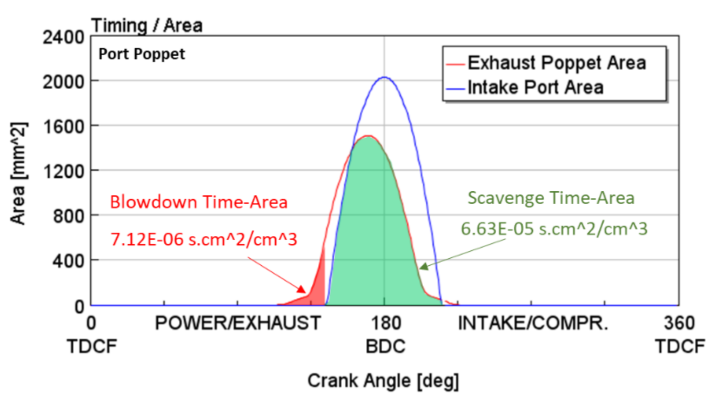


Fig. 17: Port area profiles optimized for lowest NSFC at operating point B (peak torque) for the port-poppet uniflow engine.

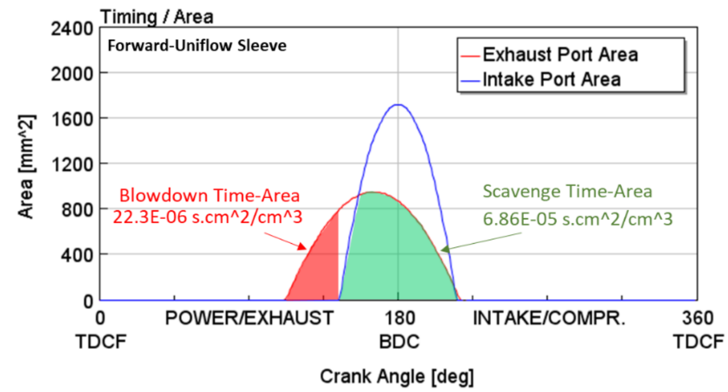


Fig. 18: Port area profiles optimized for lowest NSFC at operating point B (peak torque) for the forward-uniflow sleeve-valve engine (Rolls-Royce Crecy-type).

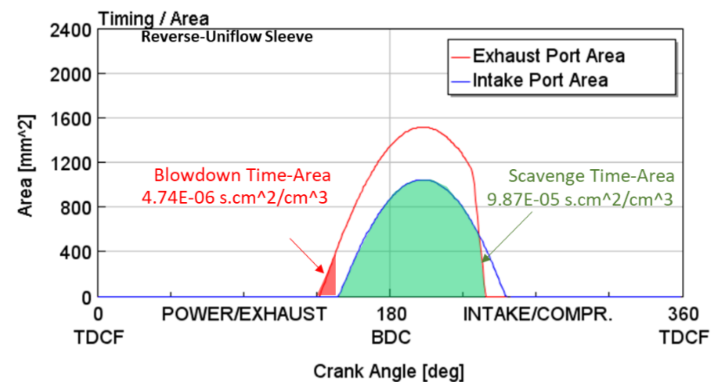


Fig. 19: Port area profiles optimized for lowest NSFC at operating point B (peak torque) for the reverse-uniflow sleeve-valve engine.

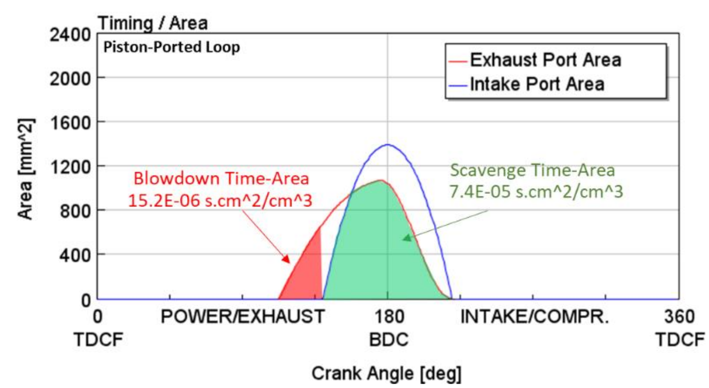


Fig. 20: Port area profiles optimized for lowest NSFC at operating point B (peak torque) for the Scnürle loop-scavenged piston-ported engine (utilizing a Lotus-type charge trapping valve to achieve asymmetric port timing).

Table 3: Numerical summary of port timings of the different concepts.

Engine Type	Optimized Valve / Port Timings [°ATDC]			
	EPO	EPC	IPO	IPC

OP2S	140	220	147	228
Port-Poppet	115	225	145	215
Forward-Uniflow Sleeve	113	218	145	216
Reverse-Uniflow Sleeve	137	239	149	252
Piston-Ported Loop	113	220	140	220

Table 4: Numerical summary of specific time-areas of the different concepts.

Engine Type	Specific Time Areas (all at 1500 rpm) [s.cm^2/cm^3]		
	Intake	Blowdown	Scavenge
OP2S	11.8E-05	2.78E-06	9.99E-05
Port-Poppet	12.6E-05	7.12E-06	6.63E-05
Forward-Uniflow Sleeve	11.9E-05	22.3E-06	6.86E-05
Reverse-Uniflow Sleeve	10.4E-05	4.74E-06	9.87E-05
Piston-Ported Loop	10.9E-05	15.2E-06	7.4E-05

From Table 2 it is readily apparent that the OP2S engine is the best configuration with regards to fuel consumption: it has the lowest values of NSFC for all three operating points, and indeed is the only one below 200 g/kWh at points B and C. It also has the lowest estimated supercharger power requirement. To reinforce this the data of Table 2 is averaged in Table 5 and then compared to the loop-scavenged engine (the worst performer) in terms of percentage change. This is done for both NSFC and the estimated supercharger power requirement, but it must be noted that this value reflects only the 'make-up' power (per cylinder) that a supercharger would have to supply as part of a compound charging system within the modelling assumptions discussed earlier.

Table 5: Comparison of results for the different scavenging concepts studied in terms of averages and change relative to the loop-scavenged piston-ported engine.

Engine Type	NSFC		Estimated Supercharger Power Requirement	
	Average of three operating points [g/kWh]	Change relative to Loop-Scavenged Piston-Ported [%]	Average of three operating points [kW]	Change relative to Loop-Scavenged Piston-Ported [%]
OP2S	188	-9.6	0.79	-45.9
Port-Poppet	204	-1.9	1.51	+3.4
Forward-Uniflow Sleeve	206	-1.0	1.50	+2.7
Reverse-Uniflow Sleeve	201	-3.4	1.63	+11.6
Piston-Ported Loop	208	-	1.46	-

Finally, while the reasons for the results of the individual concepts are discussed in detail below, with regards to the discussion above on the possibility of applying Miller-type operation to a 2-stroke engine, Table 6 presents a numerical summary of the CRs and ERs and the ratios between them.

Table 6 Numerical summary of the effective (or trapped) compression and expansion ratios of the different concepts, and the ratios between them.

Engine Type	Effective Compression Ratio [:1]	Effective Expansion Ratio [:1]	Ratio of Expansion to Compression Ratios [-]
	Volume at start of compression / clearance volume	Volume at end of expansion / clearance volume	Expansion ratio / compression ratio
OP2S	13.68	13.67	1.00
Port-Poppet	13.79	11.18	0.81
Forward-Uniflow Sleeve	13.85	11.49	0.83
Reverse-Uniflow Sleeve	10.97	13.53	1.23
Piston-Ported Loop	13.73	11.49	0.84

Here it can be seen that the reverse-uniflow sleeve-valve engine has the best value of the CR/ER metric, meaning that it is effectively an engine operating on the Miller cycle. Of the others, the OP2S is closest, with all of the other three scavenging systems then being approximately the same.

Figure 21 presents a breakdown of the power flow from the cylinder at the point B condition (or peak torque), which is useful for a visual comparison of pumping work and heat losses. However, please note that while the conditions are optimized for lowest NSFC at this point, this is *not* an efficiency breakdown.

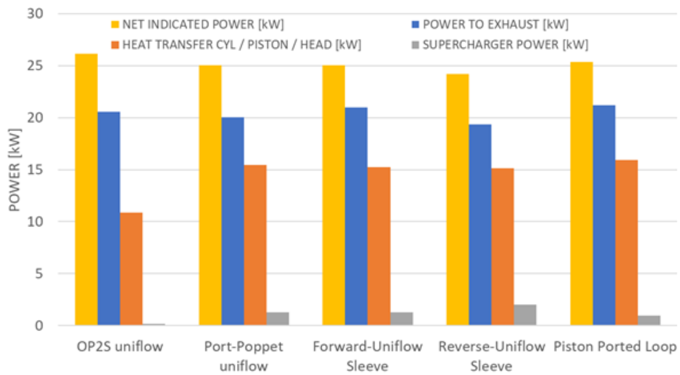


Fig. 21: Breakdown of power flow at operating point B (peak torque – 1500 rpm, 14 bar IMEP and 2.0 bar exhaust back pressure). (Note that this is *not* an efficiency breakdown – see text.)

The fact that from Table 2 the work conducted here shows the OP2S engine to be the best of the options modelled is considered to be some validation of the previous work by Warey *et al.* [38,39] in which in terms of fuel consumption the OP2S configuration was found to be better than loop scavenging and even better still than the comparable 4-stroke engines that they modelled; consequently the present work is also considered to support the work of Achates Power in terms of diesel OP2S fuel consumption advantages over current 4-stroke engines.

Discussion

This section will discuss each case individually before drawing broad comparisons between the systems.

Opposed-Piston (OP2S)

From Table 2 it can be seen that the OP2S engine delivers the lowest NSFC over all three selected operating points, and from Table 5 it is evident that it gives easily the lowest NSFC, being 9.6% lower than the piston-ported loop-scavenged engine. This result is due to a combination of the lowest ISFC coupled to the lowest supercharger work; this was 45.9% lower than the piston-ported engine. This is achieved through two principal routes:

1. Maximized expansion work. The mechanical arrangement permits later EPO which maximizes expansion work. In Table 6 it can be seen that the ER to CR ratio is 1.00, compared to values of less than 1 for all of the other concepts except the reverse-uniflow sleeve-valve (i.e. they operate under-expanded). This design therefore has this thermodynamic advantage over most of the other configurations.

2. Reduced heat transfer. Although the specific design rules applied result in a larger total surface area compared to the port-poppet and sleeve design (approximately 4.6% greater: see Table 1), the average temperature of the combustion chamber surface is higher (because the two pistons run hotter than a cylinder head would). As a consequence heat transfer is reduced. The advantage that the OP2S has in this area is clearly shown in Figure 21, where the magnitude of the power loss to heat transfer is the lowest; this was the case for all of the operating points investigated here.

The latter point is in line with what other researchers have discussed [12] but here attention is drawn to the fact that the reason is a net gain from the summation of the interaction of surface areas, heat transfer coefficients (HTCs) and surface temperatures rather than being merely a simple observation that a cylinder head is not present like in other types of engines. It is believed that this advantage could be further improved by optimization of the stroke:bore ratio. Nonetheless, as it stands, the reduced power rejected to coolant would be expected to give a benefit at the vehicle level as well.

For the OP2S the optimizer chose an IPO very shortly after EPO, resulting in the smallest exhaust blowdown period of all the concepts. However, note that the subsequent scavenge period (in terms of scavenge time-area) is the highest. It is thought that this, combined with a slightly more favourable scavenge profile, results in similar levels of trapped residual gases to the other concepts.

In the model, the short blowdown period results in a flow of residuals into the intake system at IPO due to the cylinder pressure conditions. To some extent this is observed in some of the other designs due to optimization of port timings for minimum NSFC, however it is most pronounced in the OP2S.

This concept has the lowest supercharger work which undoubtedly contributes to the low NSFC. Further retardation of timing events does reduce ISFC but then requires higher supercharger work since the incoming air has to be compressed to a greater pressure, and thus the NSFC increases.

Given the simplified nature of the charging system model (which relies heavily on estimated efficiencies) it is not possible to say whether or not the supercharger could definitely be de-clutched at any of the engine operating points. However, at peak torque the estimated charging system load is very small and allowing for an optimized turbocharger match it is possible that the supercharger could be disengaged fully, further improving fuel consumption.

The optimum phase lag between the pistons was found to be an exhaust lead of 7.5°. While is this value is specific to the geometry modelled it is not very different to the 8° value settled upon for one of the Achates Power engines [10]. After the main optimization phase of the project had been completed, the phase offset between the pistons was investigated in more detail. This was done in order to investigate what effect a theoretical device that could vary the phase between the crankshafts might have on economy. Consequently for this investigation the port geometry was fixed and the CR was allowed to vary with the change in exhaust piston lead. The only parameter that did vary during this investigation was the injection timing in order to maintain the same SOI relative to the effective volume-based TDC, i.e. all of the hardware was essentially fixed.

The results of this sweep are shown in Figure 22 for the three different loads investigated. There was found to be small benefit in ISFC and NSFC from varying piston phase for the part-load and full-power operating points, but overall the general response is remarkably flat, especially between 2.5° and 12.5° of exhaust lead. In reality, in terms of combustion control a potentially important advantage of such a complication would be the ability to provide VCR to control GCI combustion more directly. The significant potential of VCR in this context has been demonstrated by Turner *et* *al.* [7,8]. It is considered that this possibility is worthy of further investigation.

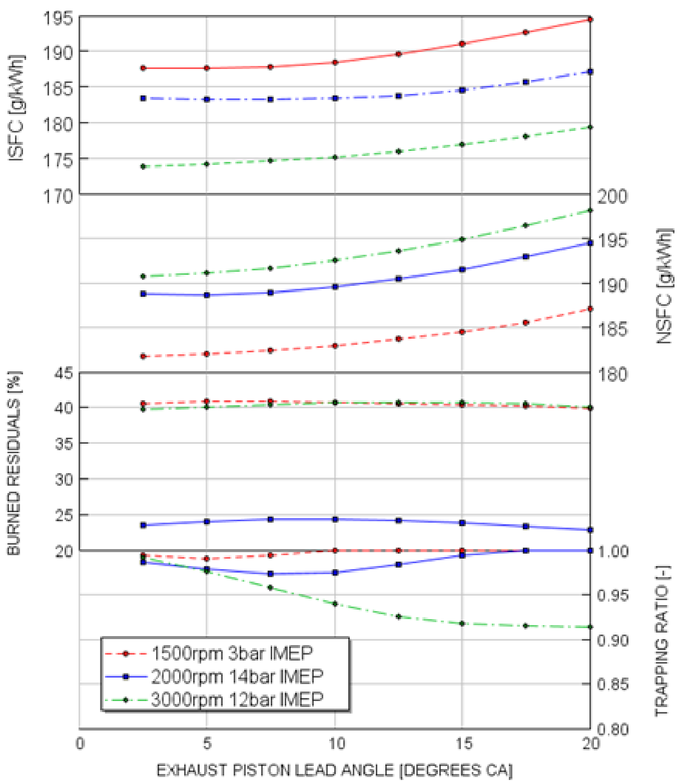


Fig. 22: Effect of exhaust piston lead angle (or piston phase) on ISFC, NSFC, burned residual rate and trapping ratio for the OP2S engine.

The reader may like to note that both the optimized value of exhaust lead obtained here and those of the Achates engines are significantly different to the value adopted by the historically-important Napier Deltic OP2S engine [44]. That engine utilized an exhaust lead of 20° not because of performance considerations but because its unusual architecture (of three crankshafts and three banks of cylinders) required it to work from a mechanical and firing interval point of view. This limitation was not shared with all other multi-bank OP2S engine such as the four-bank (and four-crankshaft) Junkers Jumo 223 and 224 engines [27,28,44].

Port-Poppet

The average NSFC is higher for the port-poppet engine versus the OP2S, but still 1.9% lower than the piston-ported engine. It is constrained by acceleration forces in its valvetrain. The profiles used here were taken from a model of a modern 2.0 litre downsized 4-stroke gasoline engine with a maximum engine speed of 6500 rpm. Modifying the profile to double its frequency whilst retaining the acceleration limit resulted in a minimum duration of 110° with 4 mm of lift. Consequently, delaying EVO to increase ER resulted in EVC occurring later in the compression stroke and the resulting loss of charge had then to be compensated by charging system work, which was 3.4% higher than the piston-ported engine. Thus this concept cannot match the late EVO of the OP2S design and has a higher ISFC. Also, from Table 6, the ER is less than the CR, meaning a thermodynamic loss versus an ideal cycle.

If there was a way to improve the poppet valve performance (i.e. shorten the duration whilst maintaining lift), it may be possible increase expansion work and improve ISFC. Such mechanisms may include the use of desmodromic valve operation, as is used in production by Ducati, or an air-valve spring system. The latter is essentially a motorsport-only system and so is not considered viable here. Other valving systems may offer benefit, but except for the sleeve valve, these are outside the scope of this investigation.

However, when using what amounts to a conventional production valve system, there is the scope to employ variable valve timing (VVT) afforded by camshaft phasing devices. When investigating this the results showed that with nominal timing optimized for peak torque, further retardation of EVO causes a small reduction in NSFC for the part load and maximum power operating points. At the peak torque condition, retarding the timing reduces residuals, probably due to increasing the scavenge time-area, although this again increases NSFC due to higher supercharger work. It is thought that using a reverse-uniflow configuration might have some potential benefit, since applying VVT to the intake instead might then facilitate the application of Miller-cycle operation at certain operating points.

Forward-Uniflow Sleeve Valve

Because it is a fundamentally different and desmodromic mechanism, the valve kinematics issues with the port-poppet arrangement can be bypassed by the sleeve valve. This, coupled to the fact that the ports can be disposed around the cylinder periphery, means that like the OP2S the sleeve valve can provide short durations for any given angle-area requirement. However, the interaction of sleeve, piston, and port makes its optimization more difficult due to geometric constraints: for example, varying the phase of the sleeve motion relative to the piston changes the exhaust port timing at the top of the cylinder, whereas the intake timing is essentially piston-controlled via an interaction with the sleeve ports at the bottom. This effect can be seen for the Crecy in Figure 5.

The data of Table 3 shows that despite this extra degree of freedom, the optimizer converged on a set of timings very similar to the port-poppet design. As a consequence the resulting simulated NSFC and estimated supercharger work requirement is also very similar for these two concepts, as shown in Tables 2 and 5. Table 6 shows that the trapped CRs and ERs are very similar for the two, as well. This concept has the largest blowdown specific time-area of all the designs, however it has very similar scavenge time-area to the port-poppet engine.

The configuration was also considered interesting from the results gathered because it had high potential for breathing at higher engine speeds. The limiting factor in this was found to be the interaction of the piston and sleeve motions near to BDC. This was one of the reasons for investigating the reverse-uniflow: the interaction would affect the exhaust phase in that iteration, not the intake as here.

Varying the sleeve phase relative to the piston was also investigated as part of this study; this changes the exhaust timing, leaving the intake timing essentially piston-controlled. The optimum sleeve phase for the peak torque point (B) was found to be a 15° lead; however, for the part load (A) and maximum power (C) conditions a 5° to 10° lag gave a slight improvement. One can imagine current camshaft phasing devices being able to provide this functionality, depending on the exact sleeve drive mechanism.

It must be stated again that for this and the reverse-uniflow sleeve-valve case, heat transfer from the cylinder was assumed to be the same as for the other designs. There may be a small benefit to NSFC from reduced heat transfer to the liner due to the presence of the sleeve, but at present this is unknown. For both sleeve-valve configurations this would be worthy of further work were either concept to be carried forward, both in terms of theoretical analysis and, initially, motoring rig-based work.

Reverse-Uniflow Sleeve Valve

As mentioned above the investigation of this configuration was driven by earlier findings rating the Crecy design versus the port-poppet configuration [34]. While the forward-uniflow sleeve-valve optimization provided very similar results to the port-poppet, it was felt from constructing and operating the original models in that initial work that it might be possible to provide increased expansion in a reverse-uniflow version of the sleeve valve layout. Furthermore, unlike in the conventional Crecy arrangement, during scavenging any plug flow through the cylinder would also provide the advantage of leaving fresh air at the top (near any injector and spark plug position) and burnt gas at the bottom, helping to insulate the piston from combustion heat loss. It is accepted, however, that the piston might be expected to run hotter in this case.

Table 5 shows that the reverse-uniflow sleeve has the second-best NSFC of the five concepts, after the OP2S. It does, however, have the highest estimated supercharger power requirement. The reason that these two opposing results can co-exist is thought to lie in the data shown in Table 6, where it can be seen that the original hypothesis is borne out: this configuration has an ER 23% greater than its CR, meaning a significant thermodynamic advantage over the others in terms of work extraction (even the OP2S). It can thus be considered to be a *bona fide* Miller-cycle-operation engine, ironically without requiring all of the poppet-valve paraphernalia that permits existing 4-stroke engines to adopt this operating strategy. In fact, arguably it may better be termed a different form of Atkinson's "Cycle Engine", because it achieves its full operating cycle in a single turn of its output shaft (although the original such concept actually operated on a 4-stroke cycle) [46]. It is believed that this possibility is a new finding for a pure 2-stroke cycle engine.

Since the intake port is at the top it is solely controlled by the sleeve motion and by adjusting the phase of the sleeve relative to the piston the intake timing can be varied, allowing a range of values of the over-expansion to be explored. Determining the exhaust timing is more complicated due to the multiple interactions involved between the sleeve, its ports and the piston. However, this design showed lower residual rates than the others and this in itself would be worth studying in more detail since it may allow yet more efficiency to be realized from this concept as well as perhaps showing some synergies with turbocompounding.

Schnürle loop-scavenged piston-ported

As mentioned previously, the specific form of this engine that was modelled was configured with the Lotus charge trapping valve in the exhaust port because this was considered to represent the most advanced timing control device for a piston-ported engine. Consequently, it would be expected to yield a best-case baseline for the other configurations.

Nevertheless, after optimization this arrangement gave the highest average NSFC relative to all of the other concepts. The two principal factors that limit this design with respect to the others are:

1. <u>Surface area</u>. Although it has a smaller surface area due to the square stroke:bore ratio, the overall heat transfer is greater than the other designs. Transfer to liner is less but to the piston and head it is greater due to the increased bore size.

2. <u>Port geometry</u>. The exhaust and intake port dimensions are limited circumferentially since they lie on the same plane of the cylinder. To increase the area, the port height has to be increased, resulting in earlier timing, reducing the expansion ratio and the work produced compared to the other concepts.

From this work it was found that the CTV in the exhaust port does improve NSFC by allowing asymmetric timing, reducing the charge loss caused in conventional loop-scavenged engines by the symmetrical and late closure of the exhaust; without the CTV this occurs in all such engines even if they have variable port height mechanisms. This work therefore validates previous research performed by some of the authors [7,8,15,36]. The optimum point was found to be with the CTV set to give EPC at 220° ATDC; however, it still underperformed the other concepts taken to this level of optimization at all operating points, although it is interesting to note that it was only slightly worse and it did *not* have the highest estimated supercharger power requirement. Considering its overall compactness and mechanical simplicity and adjustability, the concept may still be attractive for many applications.

Comparison of the different systems

From Table 5 it can be seen that, at 188 g/kWh when averaged, the NSFC of the OP2S is significantly better than the other concepts. Indeed, the NSFC results for the other four concepts are all quite close, spanning 201-208 g/kwh (a range of 3.5%). Averaging these values together yields 205 g/kWh, meaning that the OP2S is better than the average of the rest by approximately 8.3%.

This clear advantage for the OP2S stems from several areas, as mentioned above: reduced heat transfer, increased expansion work, and reduced supercharger power requirement, the latter two of which are linked. They are an embodiment of the architectural advantage of the OP2S in that it can use the whole cylinder bore circumference for its ports (allowing for the lands between them), giving the optimizer the opportunity to achieve the necessary angle-areas combined with short durations, yielding the related maximum expansion work. Because more work is generated in the cycle, this reduces the required air mass flow and with it the charging system work requirement, providing a virtuous circle in terms of the necessary angle-areas on the intake and exhaust side.

As mentioned above, the other four concepts are relatively closely matched. However, the initial premise that the reverse-uniflow sleeve-valve configuration should be better than the others appears to be borne out, despite the fact that this configuration has the highest supercharger power requirement. The unusual finding of this work is that the reverse-uniflow sleeve-valve appears to be a 2-stroke configuration which would allow differential expansion, and with it a thermodynamic advantage (see Table 6). Of the others, only the OP2S is neutral in this respect, and again it is interesting to note that the remaining three all have very similar ratios of effective expansion ratio divided by effective compression ratio, with values in the region of 0.81-0.84.

The overall results for the loop-scavenged engine shows the remarkable capacity of the CTV to improve scavenging performance; from their publications Lotus always claimed that the CTV smoothed out the torque curve of the engines it was fitted to [7,8,15,36], and this work provides insight into why this should be so. The fact that this configuration has an average NSFC only 4 g/kWh worse than the commercially-successful (in terms of applications where fuel consumption is considered of overriding importance) port-poppet engine is considered remarkable, and a positive indictment of the CTV. It should be remembered that the simple non-CTV Schnürle loop-scavenged engine was not modelled, however; doing this would be worthwhile to provide some further context.

Of the remaining port-poppet and forward-uniflow sleeve-valve engines, the former is better for NSFC and worse for estimated supercharger power requirement. As a consequence of the assumptions made in order to conduct this study it is tempting to rank them equally.

In Table 5, the OP2S has overwhelmingly the lowest estimated supercharger power requirement. Logically, due to the more-beneficial ratio between turbine work and compressor work that would be expected for the whole scavenge air supply system, this would also suggest that the OP2S might be the best suited to turbocompounding as well. While not investigated here, this has successfully been applied to 2-stroke engines in the past [21,44], with perhaps the most famous example being the Napier Nomad [47] which, like Tresilian's X-engine mentioned above, fell victim to improvements in gas turbine engine performance. Nevertheless, it is interesting to note that the last reciprocating piston engines realistically to be considered as long-range aircraft powerplants were turbocompounded 2-stroke engines. The more recent work of Witzky and co-workers is particularly pertinent in this respect as well [48]. The application of such technology should give further-improved fuel economy and is therefore considered worthy of further investigation particularly in connection with the opposed-piston and reverse-uniflow sleeve valve engines[5]. Other technologies which would be interesting to study in this context might be the stepped-piston concept, which could replace an external supercharger while still permitting turbocharging to be applied. This has been investigated extensively by Hooper [50,51] and also Lee [52] in a compounded engine, as well as being a feature (in inverted form) of the engine of Witzky *et al.* [48].

Finally, from this work and also from an academic perspective, an important finding has been that the 'standard' guidelines for deciding the angle-area requirements of conventional crankcase scavenged 2-stroke engines, as derived by researchers at Yamaha [43], have been found to be broadly applicable to all of the 2-stroke scavenging systems studied here. As discussed above, in the present work these guidelines were used as an initial starting point for port geometry, however as the study progressed it also became clear that they were not fully optimal. Hence numerical optimization of the port/valve timings was adopted. This latter stage allowed the explicit targeting of minimum NSFC in the design; however, in some cases the resultant timings showed significant differences to the values given by the guidelines, particularly for the blowdown phase. In turn this suggests that further research into angle-area requirements would be especially useful for the ongoing study of modern 2-stroke engines.

A subjective final ranking of the scavenging concepts, based on the objective results of Table 5, is considered to be as shown in Table 7.

Table 7 Ranking of the different concepts based on the results obtained from this study.

Ranking	**Engine Type**	**Comments**
1st	OP2S	Clearly the best for NSFC and estimated required supercharger power. Also the least complicated mechanically. May be possible to apply VCR via crankshaft phasing.
2nd	Reverse-uniflow sleeve-valve	Allows differential expansion. May be better still with further optimization. Has highest estimated required supercharger power. Mechanically challenging due to current engineering knowledge.
3rd =	Port-poppet	Compromised by valve kinematics. Obvious potential to vary exhaust valve timing. Not complex.
3rd =	Forward-uniflow sleeve-valve	Mechanically challenging due to current engineering knowledge.
3rd =	Loop-scavenged piston-ported	Almost as good in NSFC as port-poppet and forward-uniflow sleeve, with better estimated required supercharger power than either. This performance was greatly helped by the use of a Lotus-type charge trapping valve. Has validated potential for VCR. Mechanically simple and understood.
6	Poppet-valve reverse-loop	Not taken to full optimization stage due to having the poorest performance in the first stage of the process.

A ranking process for manufacturing was conducted as part of the project, but this is not included here due to the two facts that this was primarily an efficiency-oriented investigation and that such a ranking is dependent on many more variables (for example, the sleeve valve manufacturing process being one complete unknown in the modern world). Nevertheless, it can still be said that the OP2S engine continues to be very attractive when compared to the others in this manner, meaning that its overall advantage can be expected to increase further.

Final points with regards to this ranking are included in the Conclusions section below.

Conclusions

Several different scavenging systems suitable for use in an automotive 2-stroke engine were compared using a 1-D engine simulation code. The simulation was carried out on a single cylinder. All configurations were subject to the same indicated power and torque targets, themselves suitable for a medium-duty application. Engine displacement, geometric CR and exhaust back pressure were all matched; however, engine stroke:bore ratios were chosen to allow a meaningful comparison, given the variation in porting arrangements.

[5] More recently a large-capacity turbocompound V24 engine has been proposed to replace the gas turbines of passenger aircraft; however, the new proposal is for a 4-stroke engine [49].

The conclusions drawn from this work were:

1. The opposed-piston configuration provides the best performance since it allows for greater expansion than all of the others (except the reverse-uniflow sleeve-valve) and the minimum heat transfer. This latter benefit occurred despite the OP2S having the highest surface area-to-volume ratio of any of the configurations tested; it was the result of the summations of the product of surface area and HTC for the individual walls of the combustion chamber. It gave an average NSFC 9.6% lower than the loop-scavenged engine, which was found to be marginally the worst performing layout.

2. Varying the piston phasing of the OP2S engine was found to have minimal effect on fuel consumption, at least within a range of 2.5° to 12.5°. From previous work where VCR was found to be a major control on an HCCI-type combustion system in a 2-stroke engine [7,8], this could be a significant benefit if such a phasing mechanism can be engineered. This is considered worthy of further work.

3. The reverse-uniflow sleeve-valve engine was second in the final ranking of performance in terms of NSFC, at 3.4% lower than the loop-scavenged configuration. This was despite having the highest estimated supercharger drive power of all the systems in the complete study. Because of this performance in terms of NSFC it is considered worthy of further work because this may reveal some potential for further-improved fuel consumption. Its novelty itself makes this a possibility.

4. The poppet-valve uniflow approach was limited by the kinematics of the valve train system. Changing to a system not limited by valve springs would help in this area, but this was beyond the scope of this project which adopted valve masses and acceleration limits along with established current 4-stroke practice.

5. The forward-uniflow sleeve-valve engine was considered to have very good potential for breathing at higher engine speeds. This was slightly compromised by piston and sleeve motion interactions towards BDC, reinforcing the rationale to investigate the reverse-flow version.

6. In order to investigate both the forward- and reverse-uniflow configurations with the sleeve-valve, layout drawings and analysis of the Crecy-type sleeve had to be undertaken, using design details gleaned from the few remaining documents pertaining to this engine (essentially, the Rolls-Royce Heritage Trust book on the subject [21]).

7. The loop-scavenged engine performed essentially as well as the port-poppet and forward-uniflow sleeve-valve engines. However, this is principally due to the adoption of the Lotus charge trapping valve device, modified for use here from Lotus's publications. As a result of its mechanical simplicity, compactness, and the familiarity of its basic arrangement, it may arguably be considered more attractive than all of the others except the OP2S.

8. Heat transfer was not modified for either of the sleeve-valve engines, which, from a modern perspective, would demand most design and developmental time being spent on them. Compared to the others, this is a challenge which expanded theoretical and rig work might help to mitigate.

9. The poppet-valve reverse-loop was not taken to a full optimization; rather the concept was eliminated early on due to its poor performance at that stage. This is not to imply that there are not other merits with this concept, such as commonality of production machinery with existing 4-stroke engines, just that in terms of an approach prioritizing the fuel consumption potential of different scavenging systems it did not perform as well as any of the others.

10. Throughout this work, regardless of the configuration being analyzed, it was found that existing experiential guidelines for port angle-area specification for loop-scavenged, piston-ported engines using crankcase compression could also be applied to them all. It is believed that this has not been demonstrated before, across such a broad variation of layout. However, even using this approach, the numerical optimizer used also allowed further improvements to be made. The paper therefore presents a fundamental comparison of scavenging systems using a new approach, providing information which has not been shown before.

Recommendations for Further Work

This work has necessarily been constrained by the requirement to limit the number of variables in the analysis. It is recommended that for the most desirable configurations a sensitivity analysis be conducted to gauge the effect of different scavenging characteristics, stroke:bore ratios, port timings, pumping work, heat transfer, and the combustion rates and phasings for the GCI combustion, to allow some assessment of the accuracy of the results. The investigation of factors affecting heat transfer is very important should further analysis of the sleeve-valve configurations be carried out, for simplicity those having been assumed to be neutral with respect to the other arrangements here. Multi-cylinder engine configurations should be investigated too.

Although the comparison reported here is based on net indicated results, it would also be useful to show any sensitivity to the effect of frictional losses. This is particularly the case for the OP2S engine where the crankshaft timing mechanism will potentially introduce a significant penalty and where the frictional losses associated with the two crankshafts will be different as a result of the exhaust piston producing more power than the intake [53].

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Definitions/Abbreviations

1-D	One-dimensional
AFR	Air-fuel ratio
ATDC	After top dead centre
BDC	Bottom dead centre
BMEP	Brake mean effective pressure
BTDC	Before top dead centre
CI	Compression ignition
CR	Compression ratio
CTV	Charge trapping valve
DI	Direct injection
EPC	Exhaust port closing (timing)
EPO	Exhaust port opening (timing)
EV	Electric vehicle
EVO	Exhaust valve opening (timing)
ER	Expansion ratio
FC	Fuel cell
GCI	Gasoline compression ignition
HCCI	Homogeneous charge compression ignition
HTC	Heat transfer coefficient
ICE	Internal combustion engine
IMEP	Indicated mean effective pressure
IPC	Inlet port closing (timing)
IPO	Inlet port opening (timing)
OEM	Original equipment manufacturer
PEM	Proton exchange membrane
Px	Exhaust manifold pressure
SI	Spark ignition
TDC	Top dead centre
VCR	Variable compression ratio
VVT	Variable valve timing
$\eta_{Mechanical}$	Mechanical efficiency